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Study on Noise Reduction of High-Powered Suction Truck

Kunihiko Ishihara*[1], Michiya Okamura[2], Toshinobu Iwamoto[2], Ryo Morio[2], Mikihito Nishioka[1]

[1]Department of Clinical Engineering, Tokushima Bunri University, 1314-1 Shido Sanuki, Kagawa, Japan

[2]Kanematsu Engineering Co.,Ltd., 898-20 Myoken Nankoku, Kochi,Japan

[1]k-ishihara@fe.bunri-u.ac.jp; [2] m.okamura@kanematsu-eng.jp; t.iwamoto@kanematsu-eng.co; r.morio@kanematsu-eng.co; [1]cemikihito@kgw.bunri-u.ac.jp

Abstract

A high-powered suction truck has often been used at night in order to avoid traffic confusion in major cities in Japan. The noise reduction of the high-powered suction truck has been required from the environmental point of view. Root blowers have been widely used in order to make strong vacuum of a powerful suction vehicle. However, the noise from the end of complex pipe line becomes large. Then the silencer has usually been connected downstream from a root blower in order to reduce the noise. However, the noise of the last end is still loud. In this study, it was clarified that the composition elements located downstream of the root blower caused resonance by measuring the sound pressure and the 27.1dB reduction could be obtained by adding an insertion silencer to a cooling-water catcher. In this paper , the noise reduction mechanism will be clarified by BEM analysis and the silencer theory. As a result, the noise reduction was achieved by the following mechanism that the resonant type silencer was constructed by inserting the inner silencer to a cooling-water cacher to some extent.

Keywords

Noise Control, Silencer, High-Powered Suction Truck, Resonance

Introduction

A high powered-suction truck used cleaning of side ditch has been operated at night in order to avoid traffic confusion in major city[1]. It is necessary to reduce the noise to be required silence at night. However, a root blower makes a strong pulse in a piping and it is said that the root blower is not suitable in the noise reduction. The silencer has usually been set the downstream of the blower. Because it has been considered that the noise source of the suction truck is a final exit of the piping. In this study, the noise from the final exit of the piping is measured and analyzed the frequency spectrum. As a result, it was clarified that composition elements located downstream of the root blower became the cause of the resonance. Then the generating position of the resonance is searched and various countermeasures are carried out by trial and error. One of the trials is that inner silencers with various insertion length (0～520mm) are applied to the final exit of a cooling water catcher. As a result, the noise reduction effect of 27.1dB could be obtained at the frequency of 150Hz when the silencer's insertion length was 470mm. This was presumed that the drastic noise reduction could be obtained by avoidance the resonance due to inserting the inner silencers. Then in this study, it will be discussed that the presumption is proved by obtaining the frequency characteristics of the noise and the sound pressure distribution inside the exit tube including the inner silencer by the software "WAON".

Explanation of High-Powered Suction Truck and Its problem

Fig.1 shows the outline of suction part of a high-powered suction truck used in this study. This machine suctions salvaged materials by negative pressurizing inside the tank due to operating the root blower by utilizing the truck engine power. Namely, this is a big vacuum cleaner. The receiver tank has two wet dust separators (Primary and Secondary) to recover fine dusts. One of two wet separators (Secondary wet separator) is set the most downstream and doubled with the cooling tank for the root blower. According to the past investigations[2-4], it was clarified that the largest noise source was the sound from the final exit of the piping. Fig.2 shows the result of the sound source detection by the sound intensity method (LMS Sound brush by Siemens PLM Software Co. LTD). From this figure, it can be seen that the sound from the final exit of the piping is dominant. The present suction truck has a large overall

sound of 110dB(A) at the final exit at the rated rotating speed (1500rpm) and the frequency component of 150Hz was dominant as shown in Fig.3. We thought that the resonance generated in the piping system composed of a silencer, a NO.2 wet dust and a final exit of the piping. In the next chapter, the process of the research will be shown.

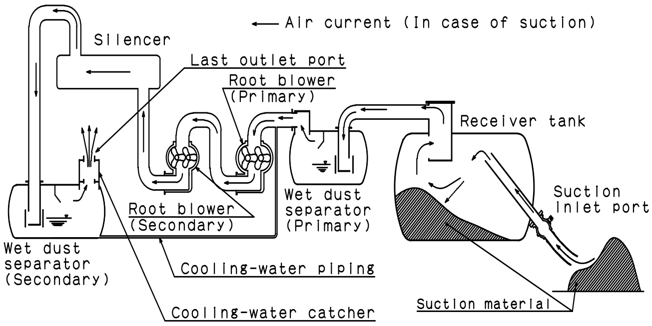


FIG.1 STRUCTURE OF SUCTION HIGH=POWERED SUCTION TRUCK

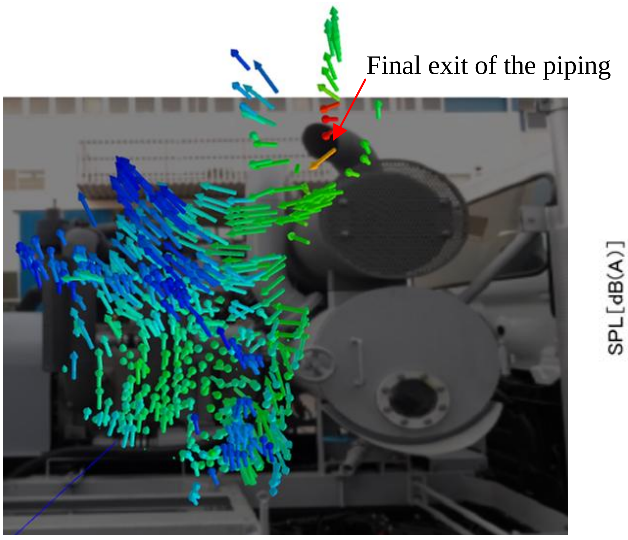


FIG.2 ACOUSTIC INTENSITY MAP OF HIGH=POWERED SUCTION TRUCK

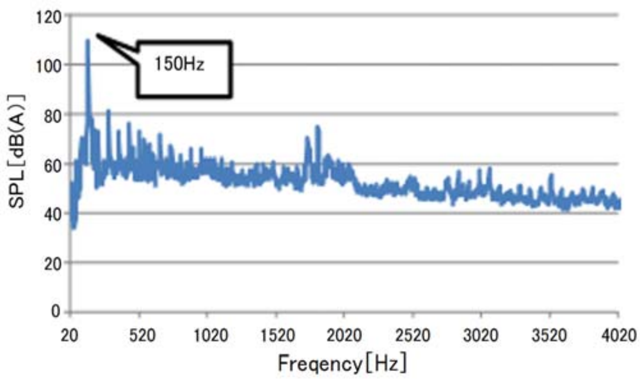
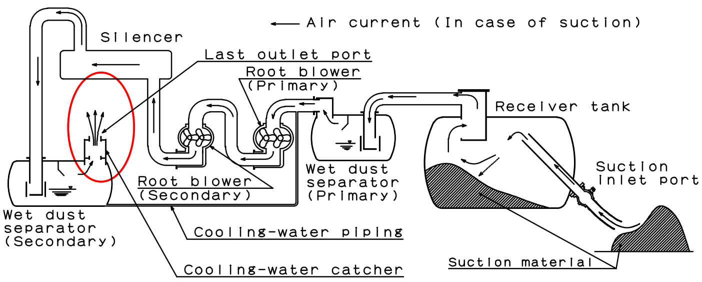


FIG.3 SPL TO FREQUENCY (STANDARD)

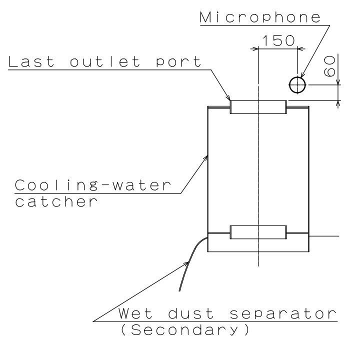
Experimental Equipment and Experimental Method

Measuring Devices and Measuring Points

The experimental equipment is shown in Fig.4(a) and the detail of the final exit of the piping, which is the noise source in this study, is also shown in Fig.4(b). A microphone is set at the position as shown in Fig.4(b). Fig.4(b) shows measuring point of the sound from the final exit of the piping system. The dominant frequency can be obtained by the expression (1). Here f is the pulsating frequency of the blower. n is the number of lobes and Z is the number of the rotor. N is the rotating speed (rpm).



(a) Experimental equipment



(b) Detail of final exit

FIG.4 MEASUREMENT POINT

$$f = nZ\frac{N}{60} \tag{1}$$

In this case, Z=2, n=3. Then Eq. (1) is rewritten by Eq. (2).

$$f = \frac{N}{10} \tag{2}$$

The measurement was basically carried out at 1500rpm. This is the rated rotating speed. We can obtain 150Hz as the pulsating frequency by expression (2). This value coincides with the dominant frequency of the sound from the final exit of the piping system. So the sound source can be said to be the root blower.

Effect of Resonance to Sound From Exit

The sound is measured by varying the blower rotating speed from 1250rpm (Minimum) to 1750rpm (Maximum) in every 50rpm. We can obtain the relation between the blower pulsating frequency and its component of sound, namely resonance curve.

Effect of Silencer to Sound from Exit

One expansion chamber type silencer used in this machine is shown in Fig.5 and its sound reduction effect *ATT* is given by Eq. (3). Where *f* is the frequency (Hz) and *c* is the sound speed (m/s).

$$ATT = 20\log m_{23} + 20\log\left|\sin kl\right| - 20\log\left|\cos kl_a\right| - 20\log\left|\cos kl_c\right| \tag{3}$$

$$k = \frac{2\pi f}{c} \tag{4}$$

$$m_{23} = \frac{S_2}{S_3} \tag{5}$$

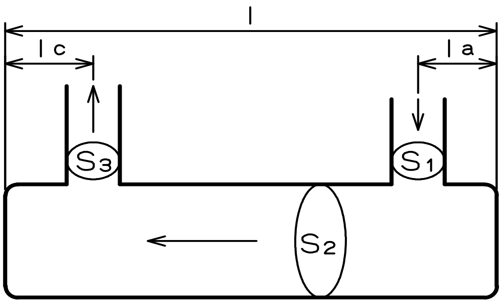


FIG.5 STRUCTURE OF SILENCER CONNECTED WITH HIGH=POWERED SUCTION TRUCK

TABLE 1 EVALUATION BY CALCULATION OF TWO KINDS OF SILENCERS

	Standard	New
l [m]	1.28	0.64
ATT [dB]	−20.8	8.79

Table1 shows the attenuation of two silencer models (standard and new versions) calculated by Eq. (3) at

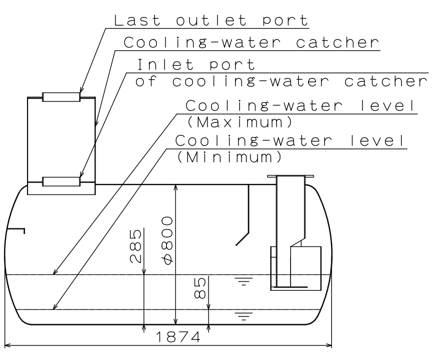


FIG.6 STRUCTURE OF WET DUST SEPARATOR (SECONDARY)

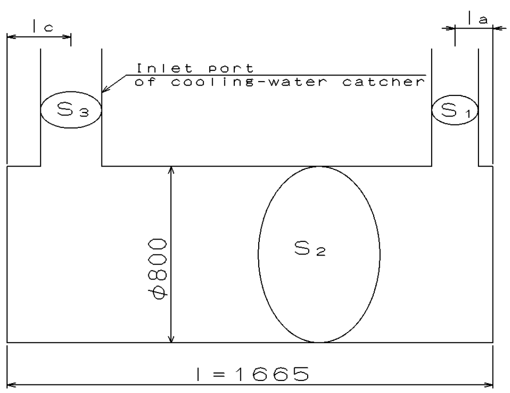


FIG.7 SIMPLE MODEL OF WET DUST SEPARATOR (SECONDARY)

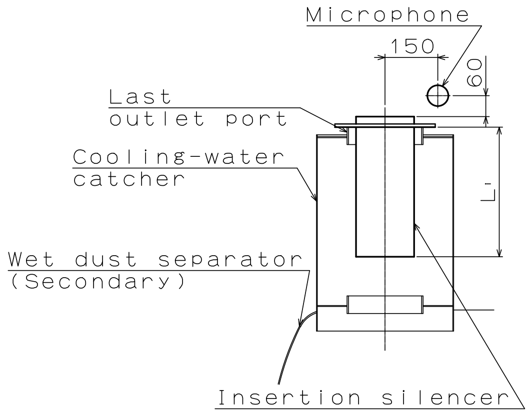


FIG.8 INSERTION SILENCER

150Hz which is the frequency of the problem. From this result, the ATT of the standard model becomes negative. Namely, the resonant occurred at 150Hz in the standard model. The sounds for two silencer models were measured and compared with each other.

Effect of Cross Sectional Area of Wet Separator to Sound from Exit

The largest composition element considered the resonance generating is No.2 wet dust separator. This equipment is shown in Fig.6. Replacing this equipment to one expansion chamber silencer like Fig.7 and applying the Eq. (3) to this simple system, the *ATT* is considered to be changed by m_{23}. Because the cooling water level changes the cross sectional area S_2 of the expansion chamber and it also makes the change of $m_{23}=S_2/S_3$.

Effect of Final Exit to Sound from Exitt

It can be seen from Eq. (3) that attenuation of one expansion type silencer depends on m_{23} described before. On the other hand, considering of a structure of wet dust separator as a silencer is shown in Fig.7, it can be thought that the cooling water catcher (Fig.6) is an exit piping of the silencer. We can obtain a large attenuation of the wet dust separator by enlarging m_{23} due to decreasing the flow pass area(S_2) of the wet dust separator. Then an inner silencer is inserted to the final exit as shown in Fig.8 and the sound is measured by varying *L*. And in *L* at which the sound becomes the smallest, the blower rotating speed is varied and the sound pressure level is measured.

Experimental Result and Its Consideration

Effect of Resonance to Sound from Final Exit

Fig.9 shows the frequency response of the sound in case of without inner silencer. The greatest difference of 16.4dB can be seen from this figure and the sound pressure level becomes conspicuously large around 160Hz. The fact means that the resonance occurs at 160Hz.

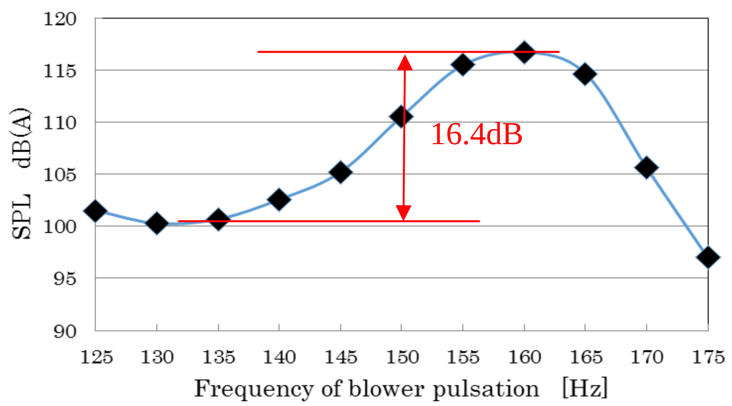


FIG.9 SPL TO FREQUENCY OF BLOWER PULSATION

Effect of Cross Sectional Area of Wet Dust Separator to Sound from Final Exit

Fig.10 shows the relation between the area ratio m_{23} of the exit piping and the expansion chamber, and the sound pressure level of 150Hz component. It can be calculated that the effect of the cross sectional area of wet dust separator to the sound from the final exit. Because the difference of only 2.8dB(A) can be seen at m_{23}=10 and m_{23}=14. m_{23}=10 and 14 are values when the cooling water level is the minimum and the maximum, respectively.

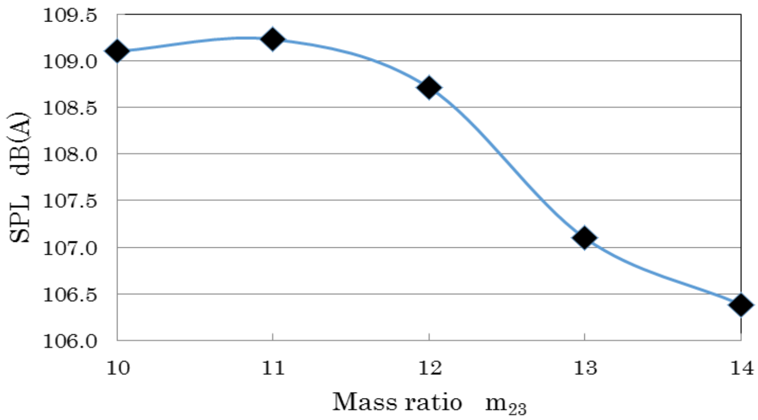


FIG.10 SPL(150HZ) TO PIPING AREA RATIO

Effect of Inner Silencer Insertion Length to Sound from Final Exit

Fig.11 shows the relation between the inner silencer insertion length L_i and the sound pressure level of 150Hz component. From this figure, it can be seen that the sound pressure level becomes minimum at a certain length L_i, in this case L_i=470mm and it could be seen that a large difference of sound pressure level, 27.1dB was obtained by inserting the inner silencer with appropriate length. From this fact, it could be seen that the resonance depends on the configurations of final exit of piping and the cooling water collector device. And it could be seen that the noise reduction effect increased with the increase of the inner silencer insertion length L_i. However, the noise reduction reveres over a certain length of L_i. Namely, the optimum length exists.

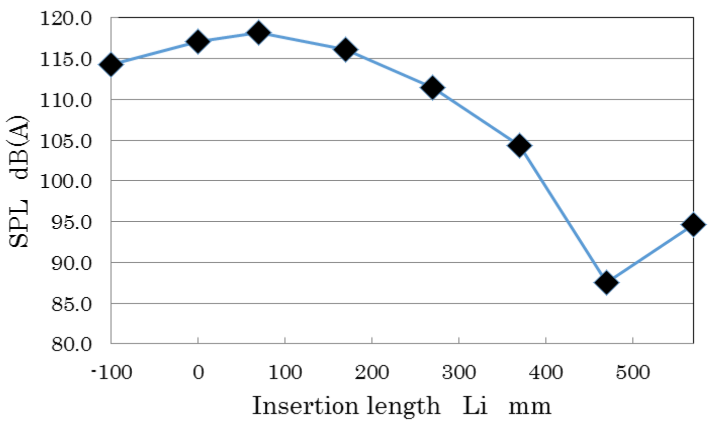


FIG.11 SPL(150HZ) TO INSERTION LENGTH OF INNER SILENCER

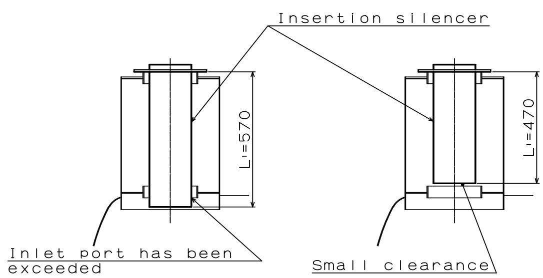


FIG.12 TWO STATES OF INNER SILENCER

This cause can be considered from Fig.12 that when L_i becomes over 470mm, the inner silencer enters the exit of the cooling water collector and the boundary condition varies largely and the sound pressure distribution of the second wet dust collector equipment including the final exit varies largely as a result. Fig.13 shows the frequency responses of sound pressure level in cases of without inner silencer and inner silencer insertion length L_i=470mm. It can be presumed that the resonance is avoided by shifting the resonant frequency. Next, this presumption will be proved by calculating the sound pressure distribution inside the final exit of the piping.

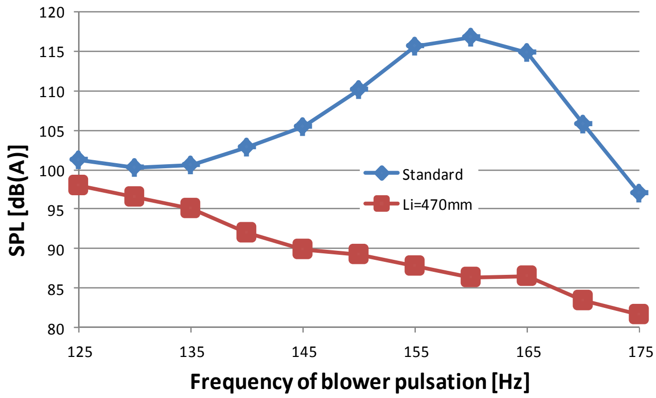


FIG.13 SPL TO FREQUENCY OF BLOWER PULSATION

Interpretation of resonance Avoidance Phenomenon by FEM

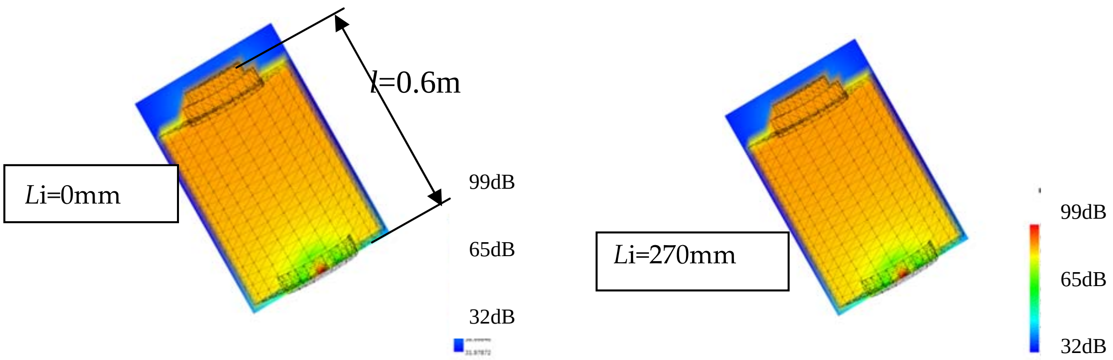
Fig.14 shows the sound pressure distributions of inside the final exit in cases of inner silencer length L_i=0mm, 270mm, 370mm, 470mm and 520mm. These results can be obtained by FEM analysis. The temperature inside the final exit is 98℃ and the sound speed becomes 386m/s by expression (6)

$$c = 331.5\sqrt{\frac{273+t}{273}} = 331.5\sqrt{\frac{273+98}{273}} = 386 m/s \tag{6}$$

Assuming that the boundary condition of final exit is one end open, and the other is closed, the fundamental resonance frequency becomes 160Hz by expression (7)

$$f = \frac{c}{4l} = \frac{386}{4 \times 0.6} = 160 Hz \tag{7}$$

Where l is the length of the exit part as shown in top of Fig.14. It can be seen that the sound pressure distribution at L_i=0mm is the resonance mode of 1/4 wave length. When the insertion length of inner silencer increases, the sound pressure distribution at L_i =370mm changes abruptly to that at L_i =470mm. That is to say, the sound field of the final exit is kept the 1/4 wave length mode from L_i =0mm to L_i =370mm and it can be considered the boundary conditions are the same in these inner silencer insertion length L_i. However, the inner silencer is inserted more deeply than Li=370mm, the sound field of final exit combines that of inner silencer. As a result, it can be changed that the sound field of the final exit is 1/4 wave length mode and that inside the inner silencer is 1/2 wave length mode (first mode).



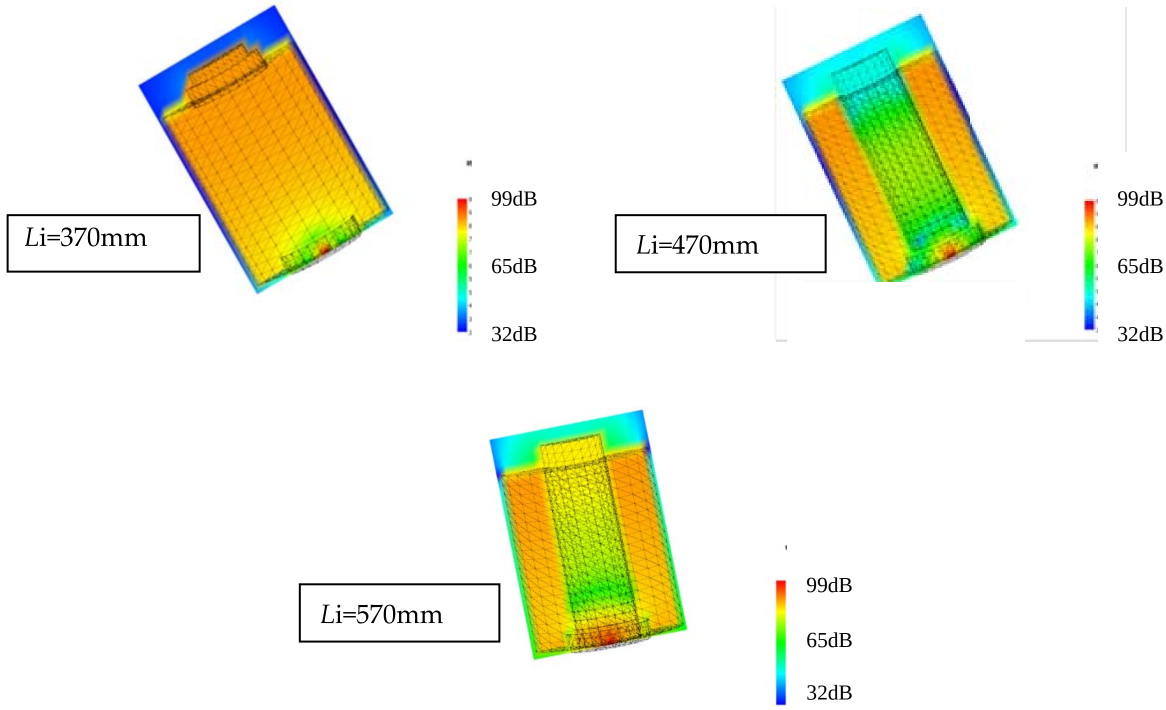
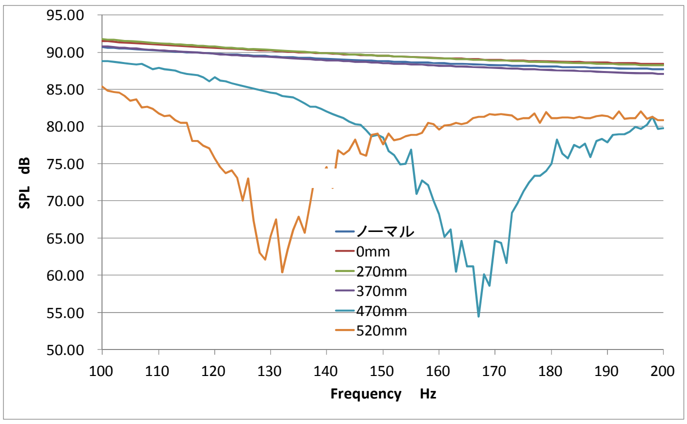


FIG.14 SOUND PRESSURE DISTRIBUTION OF FINAL EXIT PART INCLUDING INNER SILENCER AT 167HZ

This is the resonance avoidance phenomenon due to shifting the resonance frequency.

Fig.15 shows the frequency responses of the sound pressure level at the final exit of the piping in cases of L_i=0mm (without inner silencer), 270mm, 370mm, 470mm and 520mm, respectively. As can be seen in Fig.15, the sound pressure level decreases slightly with frequency till L_i =370mm and the change of the frequency response can't be seen. On the contrary, the sound pressure level decreases drastically at 167Hz as shown in blue line in case of L_i =470mm. The sound pressure distribution at this frequency is shown in Fig.14 (d) and it is very different from (a)～(c) which are the sound pressure distribution at the same frequency. Moreover, the sound pressure level increased when L_i became 520mmL_i. The cause is that the component of 167Hz becomes large from the frequency response of L_i =520mm in Fig.15. And it can be seen that the sound pressure level in case of L_i =520mm decreases at 132Hz from Fig.15. Fig.16 shows the sound pressure distribution in L_i =520mm and 132Hz. This figure indicates the both of the inner silencer and the final exit make the independent sound field and its sound field makes the first mode inside inner silencer of which boundary condition is both end open inside inner silencer.



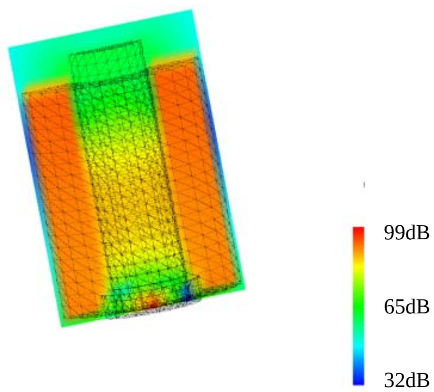


FIG.15 FREQUENCY RESPONSE OF SOUND PRESSURE FOR VARIOUS INNER SILENCER LENGTH

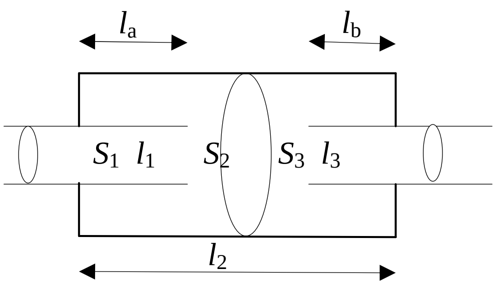
FIG.16 SOUND PRESSURE DISTRIBUTION IN CASE OF Li=520mm AND f=132Hz

Its resonant frequency can be calculated by expression (8) and it becomes 339Hz.

$$f = \frac{c}{2L} = \frac{386}{2 \times 0.57} = 339Hz \tag{8}$$

This value is very different from 132Hz which is obtained by the FEM analysis. In order to understand the phenomenon/difference, the theory of the insert type silencer is applied to this system. According to the reference [6], it is described that when the distance between two inserted pipes, ($l_2 - l_a - l_b$) becomes smaller than the diameter of S_1 and S_3, the insert type silencer changes the resonance type silencer. Then the theory will be applied to the present final exit tube and inserted type silencer.

The calculation result becomes 132Hz as shown in table2 and this value coincides with the analytical result 132Hz. From this fact, it can be seen that when the inner silencer inserted gradually, the boundary condition changes abruptly at a certain inner silencer insertion length *Li*. As a result, the resonant frequency changes and the resonant phenomenon can be avoided. Finally the large noise reduction is achieved.



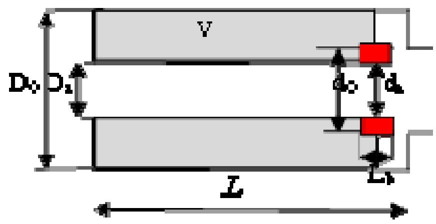


FIG. 17 FREQUENCY RESPONSE OF SOUND PRESSURE FOR VARIOUS INNER SILENCER LENGTH

TABLE 2 CALCULATION RESULT OF NATURAL FREQUENCY OF LAST OUTPUT PORT WITH INNER SILENCER

L	**Do**	**Di**	**do**	**di**
0.52	0.396	0.168	0.21	0.168
Lb	**Sb**	**c**	**freq**	
0.052	0.0125	386	131.3	

Where Sb anf freq are calculated by the eq.(9) and (10)

$$S_b = \frac{\pi(do^2 - di^2)}{4} \tag{9}$$

$$freq = \frac{c}{2\pi}\sqrt{\frac{S_b}{l_b V}} \tag{10}$$

Conclusions

The noise source detection and the frequency characteristics of noise source were examined to reduce the noise of high-powered suction truck. And the drastic noise reduction was achieved by inserting the inner silencer into the final exit tube. As a result, the following concluding remarks could be obtained.

(1) It was clarified that the cause of high level sound from the final exit tube was that the resonance was generated by giving the effect of the configuration on the sound pressure distribution.

(2) The large noise reduction could be achieved by inserting the inner silencer into the final exit tube. This is due to the resonant avoidance phenomenon which is caused by changing the silencer type, namely from the insertion type to resonance type.

(3) When the inner silencer is inserted into the final exit tube, the noise reduction effect increases with the increase of the inner silencer insertion length. However, the noise reduction effect decreases over a certain length *Li..*

(4) When the resonant place is detected in complex piping system, it is the practicable method to confirm that the high level sound decreases drastically by changing the boundary conditions of each composition element.

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Kunihiko Ishihara was born in 1947 in Kurashiki City, Okayama Prefecture Japan. He received the B.S. degree from The Kobe University in 1969. He earned the Ph.D. degree in Engineering from The Osaka University in 1986.

He worked in Kasawaki Heavy Industry Co. Ltd. as an Mechanical Engineer for 33 years. After that he became a Professor of The University of Tokushima in 2004. He had been studying the vibration and noise control, above all he studied the flow induced vibration and noise problems. He has authored or coauthored over 90 techical journal and over 40 conference papers. He is a fellow of JSME (Japan Society of Mechanical Engineers) now. He is a Professor of Tokushima Bunri University. He teaches a mechanical field for students.